

MECHANICAL ENGINEERING

GÉNIE MÉCANIQUE

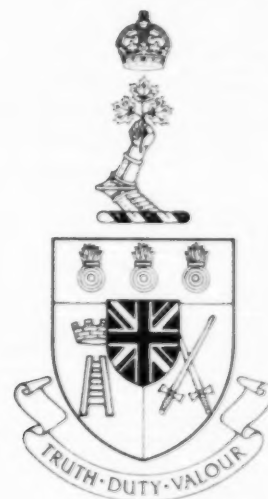
CONTROL OF SHIFT DISTURBANCES IN THE 64-RATIO KER-TRAIN TRANSMISSION

Prepared for

Transportation Development Centre

Transport Canada

June 1986



ROYAL MILITARY COLLEGE

OF CANADA

**CONTROL OF SHIFT DISTURBANCES
IN THE 64-RATIO KER-TRAIN TRANSMISSION**

BY

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June 1986

DISCLAIMER

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ACKNOWLEDGEMENTS

The authors are grateful for the the financial support of the Department of National Defence (FE 320 F 382) and the Transport Development Centre of Transportation Canada.

LANGUAGE NOTICE

Un sommaire en français de ce rapport est inclus avant la table des matières.

Transport
CanadaTransports
Canada

Development Développement

REPORT DOCUMENTATION FORM

1. Transport Canada Report No. TP 7965E	2. TDC Project No. 6570	3. Recipient's Catalogue No. N/A	
4. Title and Subtitle Control of Shift Disturbances in the 64-Ratio Ker-Train Transmission		5. Report Date June 1986	
		6. Performing Organization Report No.	
7. Author(s) J.R.G. Morrison and R.J. Ferguson		8. Transport Canada File No. D 1465-385-4	
9. Performing Organization Name and Address Department of Mechanical Engineering Royal Military College of Canada Kingston, Ont., K7K 5L0		10. DSS File No. N/A	
		11. DSS or Transport Canada Contract No. TDC 6570	
12. Sponsoring Agency Name and Address Transportation Development Centre (TDC) 200 Dorchester Blvd., West Suite 601, West Tower Montréal, P.Q. H2Z 1X4		13. Type of Report and Period Covered	
		14. Sponsoring Agency Code TDC 6570	
15. Supplementary Notes Also published as SAE 861 288		16. TDC Project Officer J.H. Morgan	
17. Abstract The transient behaviour of the gear ratio in a 64-ratio transmission was recorded for single ratio-step shifts. The transmission was installed on a test bed and operated at an input torque of 27.1 N.m. at 1500 rpm. It proved to be possible to eliminate shift disturbances entirely by microprocessor control of clutch closure times. Essentially this amounted to delaying the start of clutch closure for fast-acting clutches so that all clutches ceased slipping at the same time.			
18. Key Words Ker-Train, Automobile, Automatic, Clutch, Transmission, Microprocessor		19. Distribution Statement Limited number of copies available from Transportation Development Centre (TDC).	
20. Security Classification (of this report) None	21. Security Classification (of this page) None	22. No. of Pages 25	23. Price -

FORMULE DOCUMENTAIRE DU RAPPORT

1. N° du rapport Transports Canada TP 7965E	2. N° de l'étude — CDT 6570	3. N° de catalogue du destinataire N/A
4. Titre et sous-titre Control of Shift Disturbances in the 64-Ratio Ker-Train Transmission		5. Date du rapport Juin 1986
		6. N° du rapport de l'organisme exécutant
7. Auteur(s) J.R.G. Morrison et R.J. Ferguson		8. N° de dossier Transports Canada D 1465-385-4
9. Nom et adresse de l'organisme exécutant Département de Génie Mécanique Royal Military College of Canada Kingston, Ontario K7K 5L0		10. N° de dossier — ASC N/A
		11. N° de contrat — ASC ou Transports Canada CDT 6570
2. Nom et adresse de l'organisme parrain Centre de développement des transports (CDT) 200 Boul. Dorchester Ouest, Suite 601, Tour Ouest Montréal, Québec H2Z 1X4		13. Genre de rapport et période visée
		14. Code de l'organisme parrain CDT 6570
5. Remarques additionnelles Aussi publié comme SAE 861 288		16. Agent de projet — CDT J.H. Morgan

7. Résumé Des expériences ont été faites sur une transmission à 64 rapports pour en connaître le comportement durant les changements de vitesses, ceux-ci s'opérant un cran à la fois. Sur le banc d'essai, cette transmission a été soumise à un couple moteur de 27,1 N.m à 1,500 tr/min. Les résultats montrent qu'il est possible d'éliminer à l'aide d'un micro-ordinateur les instabilités dues aux écarts entre les temps de réponse des embrayages. La méthode utilisée consiste essentiellement à retarder l'action des embrayages les plus rapides de manière que, à chaque changements, on obtienne un temps de réponse égal de la part de tous les embrayages.			
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8. Mots-clés Ker-Train, automobile, automatique, transmission, embrayage, microprocesseur	19. Diffusion Le centre de développement des transports dispose d'un nombre limité d'exemplaires.
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0. Classification de sécurité (de ce rapport) Non classifié	21. Classification de sécurité (de cette page) Non classifié	22. Nombre de pages 25	23. Prix -
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SUMMARY

In any automatic transmission of the type typically used in passenger cars, disk clutches are used to change the transmission from one kinematic configuration to another. Thus, in the simplest epicyclic gear train, a disk clutch could be used to connect the ring gear to the input shaft so that the ring, the sun, the carrier, and the planet gears all rotate as a solid unit. There is no phasing problem in such a simple transmission because the shifting process only involves one element. The transmission would only have two ratios: 1:1 when the ring is attached to the input shaft and some ratio (depending on the sizes of the gears) when the ring is grounded to the transmission casing.

In the 64-ratio Ker-Train transmission the situation is much more complicated and there is definitely a need to control the operation of the clutches. All shifts involve at least two clutches and one shift involves all six clutches. Obviously, if the clutches do not complete their closure process simultaneously, the transmission may be - if only briefly - in another ratio. If this ratio does not lie between the before-shift and the after-shift ratios, then the drive-train may experience enough of a shock to be noticeable to the passengers of the vehicle.

In the experimental work described in the report, the transmission was run on a test bed at 1500 rpm and an input torque of 27.1 N·m. A computerized data acquisition system measured the instantaneous gear ratio during the shifting process. Only single increment shifts were examined and a great effort was made to ensure repeatability. In all, nearly 1000 tests were recorded.

The experimental work revealed that there was a considerable variation in clutch closure times for the six clutches in the transmission. The slowest clutch took about three times longer to close than the most rapid clutch. Consequently, when the transmission was allowed to shift without any control of the clutches, there was - in some cases, but not all - a significant shift disturbance.

In the second phase of the research, an effort was made to eliminate shift disturbances. Individual clutch closure times were measured carefully. These data were used to calculate the delay times needed by individual clutches involved in a particular shift so that all the clutches would complete the closure process together. For example, if one clutch could close in 100 ms, while the second clutch needed 200 ms, the initiation of clutch actuation for the fast clutch was delayed by 100 ms. This method proved effective and it was found that the shift disturbances could be eliminated.

From this research program it can be concluded that the shift disturbances in the Ker-Train transmission, due to unequal clutch closure times, can be eliminated. The calculations involved are trivial and they can be performed in real time without difficulty.

SOMMAIRE

Dans une transmission automatique type équipant les voitures de tourisme, ce sont des embrayages à disques qui commandent les transformations du train d'engrenages. Ainsi, dans un train épicycloïdal simple, on peut se servir d'un embrayage à disque pour solidariser en rotation l'arbre moteur et la couronne pour que l'ensemble des éléments tournants du train : couronne, planétaire, porte-satellites et satellites, tourne en bloc. Dans ce cas, il n'y a aucun problème de synchronisme de vitesse, étant donné qu'on n'agit que sur un seul élément tournant, le changement de vitesse aboutissant à l'un de deux rapports possibles : une prise directe par la solidarisation de l'arbre moteur à la couronne ou, lorsque la couronne est rendue solidaire au carter, un rapport quelconque variant selon la taille des engrenages.

Dans le cas de la transmission Ker-Train à 64 rapports de démultiplication, les choses sont beaucoup plus complexes et la régulation des temps de réponse des embrayages s'impose impérativement. De plus, tout changement de vitesse met en œuvre au moins deux embrayages et il en est un qui met en œuvre tous les six. Il est clair que, si les débrayages et les embrayages ne s'opèrent pas simultanément, le rapport obtenu pourrait être, ne serait-ce que très peu de temps, tout autre que le rapport désiré. Or, si ce rapport ne se situe pas entre celui d'avant le passage et celui d'après le passage, la transmission pourra subir un choc assez fort pour que les passagers le remarquent.

Au cours de l'expérimentation décrite dans le rapport, la transmission a été soumise sur le banc d'essai à un couple moteur de 27,1 N·m à 1 500 tr/min. Une chaîne de mesure pilotée par ordinateur a mesuré chacun des rapports instantanés résultant d'un changement de vitesse, les changements s'opérant un cran à la fois, dans un sens ou dans l'autre, le tout avec un souci constant de répétabilité. Près d'un millier d'essais ont été faits et enregistrés.

Les résultats ont montré de grands écarts entre les temps de réponse des six embrayages, le plus lent étant trois fois plus long que celui du plus rapide. Lorsque la transmission fonctionnait sans régulation des embrayages, il en résultait, parfois, un régime instable perceptible.

La phase suivante de la recherche a été consacrée à l'élimination des instabilités. Pour cela, après avoir mesuré soigneusement les temps de réponse de chacun des embrayages, des temps de retard ont été calculés pour que, à chaque changement, on obtienne un temps de réponse égal de la part de tous les embrayages. Si, par exemple, l'embrayage A met 100 ms et l'embrayage B 200 ms, l'action du premier est retardée de 100 ms. Cette méthode s'est révélée efficace, débouchant sur une élimination effective des instabilités.

Ainsi, le programme de recherche sur les instabilités dans la transmission Ker-Train dues aux écarts entre les temps de réponse a atteint l'objectif fixé. Les calculs nécessaires sont des plus simples et se font facilement en temps réel.

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1. INTRODUCTION

The transmission studied for this report was designed for an electric vehicle in a research and development program sponsored by the Government of Canada. The fundamental principles of operation of the transmission, known as the Ker-Train Binary Logic Incrementally Variable Transmission, have been explained in previous publications [1,2,3]. The transmission is the invention of J.H. Kerr for which patent protection has been obtained [4].

The transmission has a 9:1 to 1:1 ratio range divided into 64 steps and accepts a maximum input torque of 67.8 N·m at 3000 rpm. The transmission functioned well in the electric vehicle; however, the road tests showed that a few of the gear shifts were significantly rougher than the others which were almost undetectable. Consequently, it was decided to investigate the feasibility of controlling the shifts to minimize or eliminate the roughness.

This report will demonstrate that the shift disturbances are due to response delays in the movement of the hydraulic pistons that actuate the clutches. The behaviour of the transmission is not an uncontrolled response. It is a predictable consequence of the unsynchronized action of the clutches and hence there is a systematic method for balancing shifts.

2. THE TRANSMISSION

The transmission consists of three modules or stages of a unique and patented arrangement of planetary gear sets and inverted gear train loops [2,3,4]. Each stage can be made to produce one of four possible ratio states including 1:1. The individual stages contribute to a geometric progression of ratios. One ratio is selected from each stage and thus the overall gear ratio (input speed divided by output speed) is

$$\mathcal{R} = r_1 \cdot r_2 \cdot r_3 \quad (1)$$

where the four possible ratios for each stage are given in terms of the basic stage ratio x which is the common ratio of the geometric progression, that is,

$$\begin{aligned} r_1 &= x^3, x^2, x^1, 1 \\ r_2 &= x^{12}, x^8, x^4, 1 \\ r_3 &= x^{48}, x^{32}, x^{16}, 1 \end{aligned} \quad (2)$$

In a 9:1 to 1:1 64-ratio transmission $x = 9^{1/63} = 1.0355$.

The ratio selection is made with disk clutches of the type found in any standard automatic transmission. The schematic diagrams in Fig. 1 and Fig. 2 show the operation of the clutches to provide the four ratios which are referred to as W , Y , Z , and 1. The first two stages of the transmission are designed according to Fig. 1 and the third according to Fig. 2. In the actual transmission the clutches in stage 3 are hydraulically actuated whereas in stages 1 or 2 clutches C_2 and C_4 are held closed by springs and only clutches C_1 and C_3 are actuated hydraulically. The clutch pairs C_1 - C_2 and C_3 - C_4 contain interactive members making it impossible for the clutches to be activated simultaneously.

3. KINEMATIC RELATIONSHIPS DURING SHIFTING

Equations for the angular velocities of the gears of the transmission for any ratio were derived in [3] and will not be repeated here. During a gear shift, however, the situation is complicated by the slipping clutches and additional equations are needed.

Each friction clutch is assumed to close according to

$$\phi = 1 - e^{\gamma t} \quad (3)$$

where $\phi = 0$ at the beginning of the shift and $\phi = 1$ at the end of the shift. γ is found from the time τ_C for which the clutch is 99.9% closed, i.e.,

$$\gamma = -\frac{\ln(1 - 0.999)}{\tau_C} \quad (4)$$

τ_C will be referred to as the closure time. It is a function of the torque on the clutch, the angular velocity of the clutch components, the coefficient of friction of the plates, the oil pressure, etc.

Beginning with stage 1 or 2 (Fig. 1) the fundamental angular velocity equation is:

$$\omega_A = k_1\omega_D + k_2\omega_C + k_3\omega_G \quad (5)$$

The constants k_i are given in [3] or they may be obtained using the standard procedures for analyzing planetary gear trains [5].

ω_C and ω_A can be defined in terms of ω_G (input) and ω_D (output) as follows

$$\begin{aligned} \omega_C &= (1 - \phi_1)\omega_G && \text{when } C_1 \text{ is slipping} \\ \omega_C &= \phi_2\omega_G && \text{when } C_2 \text{ is slipping} \\ \omega_C &= 0 && \text{when } C_1 \text{ is closed} \\ \omega_C &= \omega_G && \text{when } C_2 \text{ is closed} \end{aligned} \quad (6)$$

and

$$\begin{aligned} \omega_A &= (1 - \phi_3)\omega_D && \text{when } C_3 \text{ is slipping} \\ \omega_A &= \phi_4\omega_D && \text{when } C_4 \text{ is slipping} \\ \omega_A &= 0 && \text{when } C_3 \text{ is closed} \\ \omega_A &= \omega_D && \text{when } C_4 \text{ is closed} \end{aligned} \quad (7)$$

Using Eqns (6) and (7), Eqn (5) can be rewritten to give the gear ratio for the stage during shifting as

$$\frac{\omega_G}{\omega_D} = \frac{C_3(1 - \phi_3) + C_4\phi_4 - k_1}{[C_1(1 - \phi_1) + C_2\phi_2]k_2 + k_3} \quad (8)$$

In order to apply Eqn (8) a compendium of clutch states is required as shown in Table 1.

The third stage can be dealt with in the same way. The fundamental equation corresponding to Eqn. (5) is

$$\omega_A = k_1\omega_D + k_2\omega_G + k_3\omega_C \quad (9)$$

and the gear ratio is given by

$$\frac{\omega_C}{\omega_D} = \frac{k_1 + k_2[C_3(1 - \phi_3) + C_4]\phi_4}{C_1(1 - \phi_1) + C_2\phi_2 - k_3} \quad (10)$$

Table 1 is valid for the third stage provided that Y and Z are interchanged. For example, for the $W \rightarrow Y$ shift in the third stage, read the $W \rightarrow Z$ entry in Table 1.

4. IDEAL SHIFTS

The equations developed in the previous section were used to write a computer program to show the overall gear ratio in the transmission during shifts. Fig. 3 shows the result for single upshifts and downshifts through the entire ratio range of the transmission. The clutches have all been assumed to have equal closure times of 0.5 seconds, with all clutches starting to close at precisely the same instant. Clutches were assumed to open instantaneously. The program does allow individual closure times and delay (in starting to close) times to be set, but these features were not used for Fig. 3.

Fig. 3 shows that all the shifts are smooth except for three in each series where the ratio is temporarily reversed for part of the shift. To explain this, consider the binary representation of the gear ratio exponent. In the lowest of the sixty-four gears, the ratio is x^{63} and the binary equivalent of 63_{10} is 111111_2 . By taking '1' to mean that in a binary clutch pair such as C_1-C_2 in Fig. 1, clutch C_1 is grounded to the casing, and by taking '0' to mean a connection to the input or output shaft, the operation of the clutches is immediately obvious from the binary number for the gear. With this idea in mind, shifts that involve several clutches acting together are likely to be difficult. Thus the upshifts 110000 to 101111, 100000 to 011111, or 010000 to 001111 involving the operation of five clutch pairs are likely sources of a shift disturbance. Indeed, when Fig. 3 is examined carefully, it is those shifts that are seen to cause the trouble, namely 48 to 47, 32 to 31, and 16 to 15.

Although the shifts for Fig. 3 were assumed to take equal times, the changes in ratio in the three stages are quite different. In multiplying the individual stage ratios from Eqns (8) and (10), the product does not necessarily increase or decrease monotonically. When the higher order bits in the binary number are changed, the change in ratio is larger. Thus the 32 to 31 shift should display the largest excursion from the correct ratio as indeed it does.

5. TEST APPARATUS

Input drive power for the transmission was supplied by an 3500 rpm 15 kW induction motor combined with a 37 kW capacity eddy current coupling and automatic controller (Fig. 4). Any combination of input speed and input torque, within normal operating ranges of the transmission, could be selected. Output power was absorbed in a manually adjustable Prony brake. All test functions were managed by an IBM PC-XT operating in communication with a peripheral data acquisition and control unit.

As previously explained, the ratio state of the transmission is determined by the state of six double-acting clutch packs. Hydraulic flow to these clutches was regulated by six

solenoid valves. Hydraulic fluid was supplied at 120–140 kPa by a 2.2 kW hydraulic pump. Standard automatic transmission fluid was used as the hydraulic fluid and served as the lubricant and coolant.

Power for the solenoid valves was supplied by a twelve volt automotive battery. Power supply to the solenoid valves could be switched by addressing a power controller in the data acquisition unit that accessed six DC relays. The solenoids were rated to have a maximum of 3 ms variation in response time. Therefore, only solutions for transmission shift control which involved multiples of 10 ms delays were considered to be significant.

Torque data were collected from torque cells mounted in the drive line. The angular velocities were derived from toothed wheels incorporated in the torque cells. The transducer signals were passed through preprocessors. The input to the data acquisition unit consisted of four 1 volt signals which were proportional to the input speed, input torque, output speed, and output torque.

6. EXPERIMENTAL METHOD

As part of each test series, the transmission was operated at test loads for ten minutes, while all clutch packs were shifted repeatedly to establish normal steady state operating conditions. The accumulator precharge pressure was set to 55 kPa and the battery was fully charged. However, no monitoring of hydraulic fluid temperature, accumulator pressure or solenoid currents was performed.

For any given test of a shift, the operator specified input speed, input torque and a delay time for each clutch pack involved in the shift. When steady state was established for the selected input conditions, the test was initiated by the operator. Data for 939 tests were recorded in the experimental program.

Data were acquired at 500 Hz per channel with 12 bit resolution for a period of three seconds. Clutch control signals were output concurrently with the data acquisition functions. In the first and third seconds, no shift signals were issued. No test demonstrated clutch activity during the first or third second of data acquisition. These periods were used to determine the preshift and postshift steady state conditions. All the information related to the actual shift performance was contained in the middle second of data acquisition.

7. UNTIMED SHIFT DATA

Theoretically, with 64 ratios there are 4032 possible shifts. Since it was impossible to study all of these, it was decided to deal with single increment shifts only. These can be grouped into nine shift classes as characterized by their clutch activity patterns shown in Table 2. The letter x stands for either 1 or 0 and represents a clutch that does not change state during the shift. There are 63 single increment upshifts and 63 single increment downshifts.

Gear ratio traces for seven of these nine shift classes showing single upshifts and downshifts are in Fig. 5. The shifts were untimed and allowed clutches to close at their *natural* rate. The input torque for the tests was 27.1 N·m and the input speed 1500 rpm. No filtering of any kind was performed on these data.

Many of the shifts in Fig. 5 are smooth and proceed monotonically from one ratio to the next. In other cases, such as the 7–8 shift, there is a definite move towards an

intermediate ratio creating a shift disturbance. In a later section it will be shown that such disturbances can be removed by synchronizing clutch actuation.

Data for shift classes 8 and 9 are not reported here because, in the course of experimentation, it was discovered that the hydraulic pump and accumulator system, as presently configured, could not supply adequate hydraulic flow to the sixth clutch pack for proper operation of the transmission. A loss of continuum (loss of reactive contact) in the transmission coupled with the low output inertia and high output torques of the test drive line, resulted in a rapid deceleration of the output shaft. For this reason, the 31-32, 32-31, 47-48 and 48-47 shifts were excluded from further study. This should not, however, be construed as an insurmountable inadequacy in the transmission design. If sufficient hydraulic volume flow were available to the system, the transmission would not exhibit this characteristic.

8. DATA ANALYSIS

Characterization of shift performance was based on an analysis of the speed ratios during the shift. These curves provide a clear indication of when the no-slip condition was achieved and clutch closure can be considered complete. The slopes are an indication of the output acceleration per unit input which in turn may be analysed to determine the times when the first effects of other clutches are felt by the system.

The data were filtered with a second order Butterworth filter to eliminate random noise. Both a forward and a reverse pass through the filter were performed to eliminate the phase shift that would otherwise occur with a single pass. The cut-off frequency was taken as the input rotational frequency (25 Hz at 1500 rpm).[6] The ratio of input to output speed was produced from the filtered speed values. Each ratio was analysed for features such as overshoot and undershoot, time lag between output shift signal and actual shift initiation, and the total time of shift activity.

The preshift and postshift steady state speed ratios were analysed to determine the maximum, minimum and mean speed ratios. In all tests conducted, no test exhibited a steady state error that was greater than 0.5% of the steady state mean. This fact forms the basis of the method for the identification of shift initiation.

A shift was considered initiated when the ratio departed from the steady state average and was on a continuous slope of a curve which would eventually exceed the 0.5% deviation criterion (Fig. 6). While this method was largely successful in determining event initiation times, it cannot be entirely perfect when some random error exists in the digital data acquired. Therefore, any variation in the observed initiation and completion times will not be entirely the product of the transmission operation. Over a large number of tests, the variations should balance out and the average reported times will be a reasonably accurate reflection of the physical system.

Shift completion was analysed in a manner that is completely analogous to shift initiation.

The output torque signal was analysed to determine its maximum and minimum values during the shift. Vibrations in the test bed (not caused by the transmission) created noise on the input torque signal which could not be eliminated by filtering. The noise was so

severe that the input torque measurements were useless. Fortunately the vibrations were not present during the efficiency testing reported in [3].

9. ELIMINATION OF SHIFT DISTURBANCES

The first step in this process was to determine the operating characteristics of individual clutches for particular input conditions. The transmission was operated at 27.1 N·m, 1500 rpm input, while clutch packs 1 to 5 were shifted individually. These shifts are equivalent to the 0-1, 0-2, 0-4, 0-8 and 0-16 upshifts and downshifts. The average measured clutch closure times are as follows:

Shift	Time (ms)	Shift	Time (ms)
0-1	143.8	1-0	75.0
0-2	128.5	2-0	97.0
0-4	196.9	4-0	156.2
0-8	254.8	8-0	136.4
0-16	222.2	16-0	223.2

These individual clutch closure times were determined by at least ten tests of each shift. They proved to be very repeatable and displayed a percentage standard deviation of less than 2.5%.

To modify a shift such as 7-8, shown with filtered data in Fig. 7, the closure times of 254.8, 156.2, 97.0, and 75.0 ms for the clutches involved must be equalized. This was done by introducing delays of 0, 100, 160, and 180 ms into the shifting process to postpone activation of the fast acting clutches. The result shown in Fig. 8, with filtered data, is a dramatic improvement. The sharp drop in ratio that characterized the beginning of the untimed 7-8 shift has been totally eliminated. The time to complete the shift has been reduced from 496 ms to 219 ms.

The effect on the measured output torque was also significant. During the transient for a shift such as 7-8, the torque for the untimed case would rise to a value well above the preshift mean torque and then drop to well below the postshift mean torque. By introducing delay times these excursions were reduced to 20% and 40% of their untimed counterparts. Thus there is a direct link between torsional disturbance and speed ratio performance.

The shift balancing procedure was also tried with the 16-15 shift and similar results were obtained. As the 7-8 and 16-15 shifts represent the two worst disturbances in Fig. 5, it is highly probable that any rough shift can be made smooth by introducing the appropriate clutch delay times.

10. COMPARISON WITH MODEL

The final stage in this discussion of the control of shift disturbances is to verify that the measured unsynchronized clutch closure times for a shift such as 7-8 (cited above) will produce the correct transmission behaviour when used in the kinematic model described previously. From Fig. 9 it is clear that they do give this result. The curve of Fig. 9 does indeed exhibit the characteristic performance observed experimentally. The theoretical

curve reaches its minimum before the actual curve, however, this is because the kinematic model assumes instantaneous opening for the clutches. The results demonstrate that the nature of the problem is entirely related to the clutch closure times and that correction of shift disturbances can be achieved by proper design of the hydraulic system and the control system.

11. CONCLUSION

This work has established that smooth shifts in the Ker-Train 64-ratio transmission can be obtained by controlling clutch actuation delay times. It is important to note that this can be done with mechanical hardware (proper hydraulic design) for the most part. A microprocessor-based controller can be employed to eliminate any small errors that remain after the best mechanical design is created. Also, algorithms to compensate for variations in transmission temperature, hydraulic pressure, clutch wear, etc., could be incorporated into the controller.

Tuning of torsional disturbance is available by tuning the speed ratio directly. The extent of the reduction will depend on the nature of the vehicle in which the transmission is installed. The test apparatus had a very low output inertia compared to a real vehicle and hence only the percentage reductions in output torque cited in the report have any relevance to vehicles.

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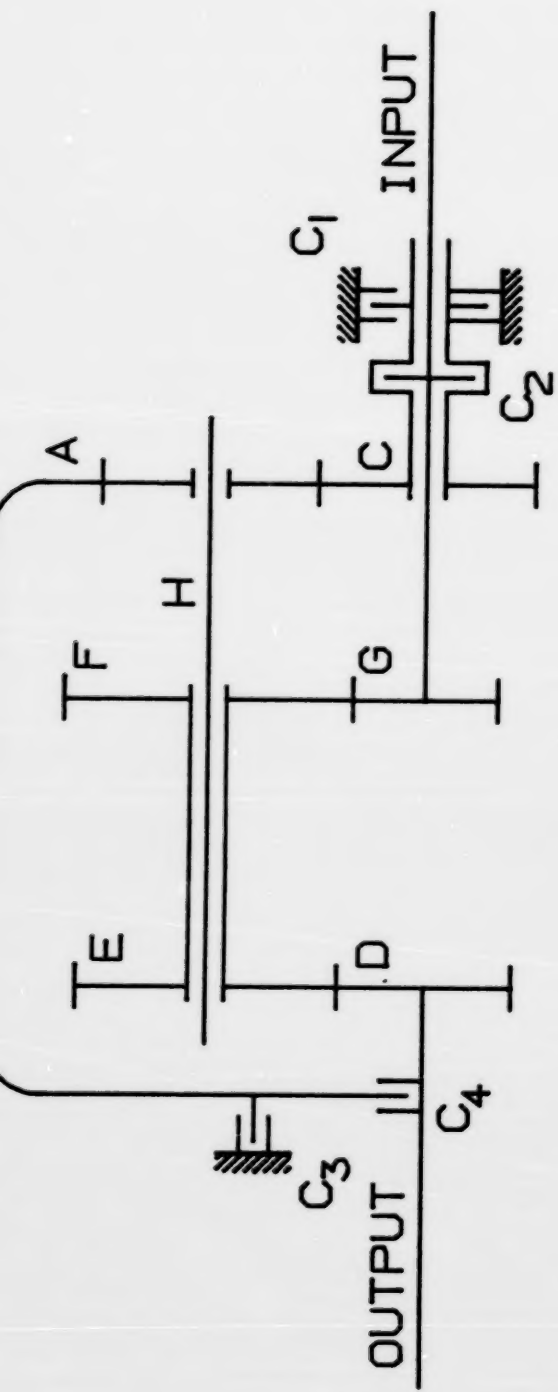
TABLE I
STAGES 1 OR 2
CLUTCH ACTIVITY TABLE

<u>SHIFT</u>	<u>SLIPPING</u>	<u>CLOSED</u>	<u>CONSTANTS</u>
W → Y	2	3	$C_2 = C_3 = \phi_3 = 1$
Y → Z	1,4	0	$C_1 = C_4 = 1$
Z → I	2	4	$C_2 = C_4 = \phi_4 = 1$
Y → W	1	3	$C_1 = C_3 = \phi_3 = 1$
Z → Y	2,3	0	$C_2 = C_3 = 1$
I → Z	1	4	$C_1 = C_4 = \phi_4 = 1$
W → Z	4	1	$C_1 = C_4 = \phi_1 = 1$
W → I	2,4	0	$C_2 = C_4 = 1$
Z → W	3	1	$C_1 = C_3 = \phi_1 = 1$
I → W	1,3	0	$C_1 = C_3 = 1$
Y → I	4	2	$C_2 = C_4 = \phi_2 = 1$
I → Y	3	2	$C_2 = C_3 = \phi_2 = 1$

NOTE: ANY C_i NOT LISTED ARE ZERO

TABLE 2

	Clutch pattern	No	Examples
1.	<i>xx xx</i> 00 - <i>xx xx</i> 01	16	0- 1, 4- 5, 8- 9,12-13...
2.	<i>xx xx</i> 01 - <i>xx xx</i> 10	16	1- 2, 5- 6, 9-10,13-14...
3.	<i>xx xx</i> 10 - <i>xx xx</i> 11	16	2- 3, 6- 7,10-11,14-15...
4.	<i>xx</i> 00 11 - <i>xx</i> 01 00	4	3- 4,19-20,35-36,51-52
5.	<i>xx</i> 01 11 - <i>xx</i> 10 00	4	7- 8,23-24,39-40,55-56
6.	<i>xx</i> 10 11 - <i>xx</i> 11 00	4	11-12,27-28,43-44,59-60
7.	00 11 11 - 01 00 00	1	15-16
8.	01 11 11 - 10 00 00	1	31-32
9.	10 11 11 - 11 00 00	1	47-48

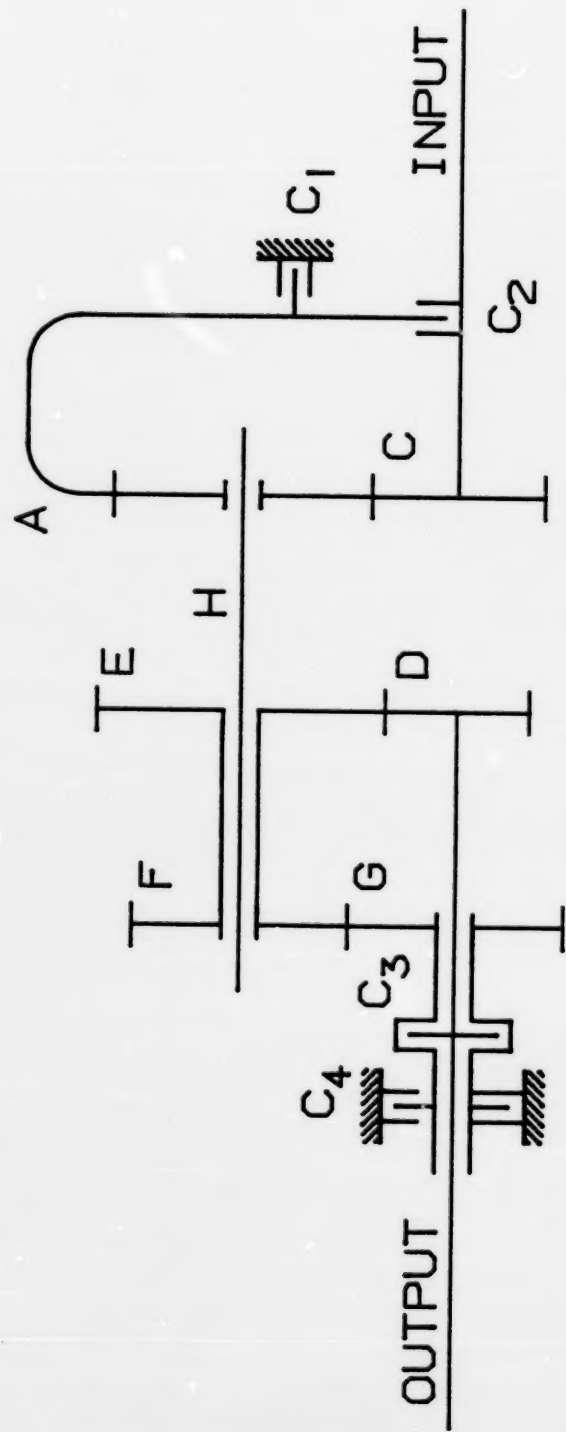


SCHEMATIC

STAGES 1 OR 2

RATIO	CLUTCHES CLOSED
W	C ₁ C ₃
Y	C ₂ C ₃
Z	C ₄ C ₁
I	C ₂ C ₄

Fig. 1 Schematic Diagram, Stages 1 or 2.



RATIO	CLUTCHES CLOSED
W	C ₁ C ₃
Y	C ₁ C ₄
Z	C ₂ C ₃
I	C ₂ C ₄

SCHEMATIC

STAGE 3

Fig. 2 Schematic Diagram, Stage 3.

IDEAL BLIncVT SHIFT PATTERNS

ALL CLOSURE TIMES 0.5 s, NO DELAYS

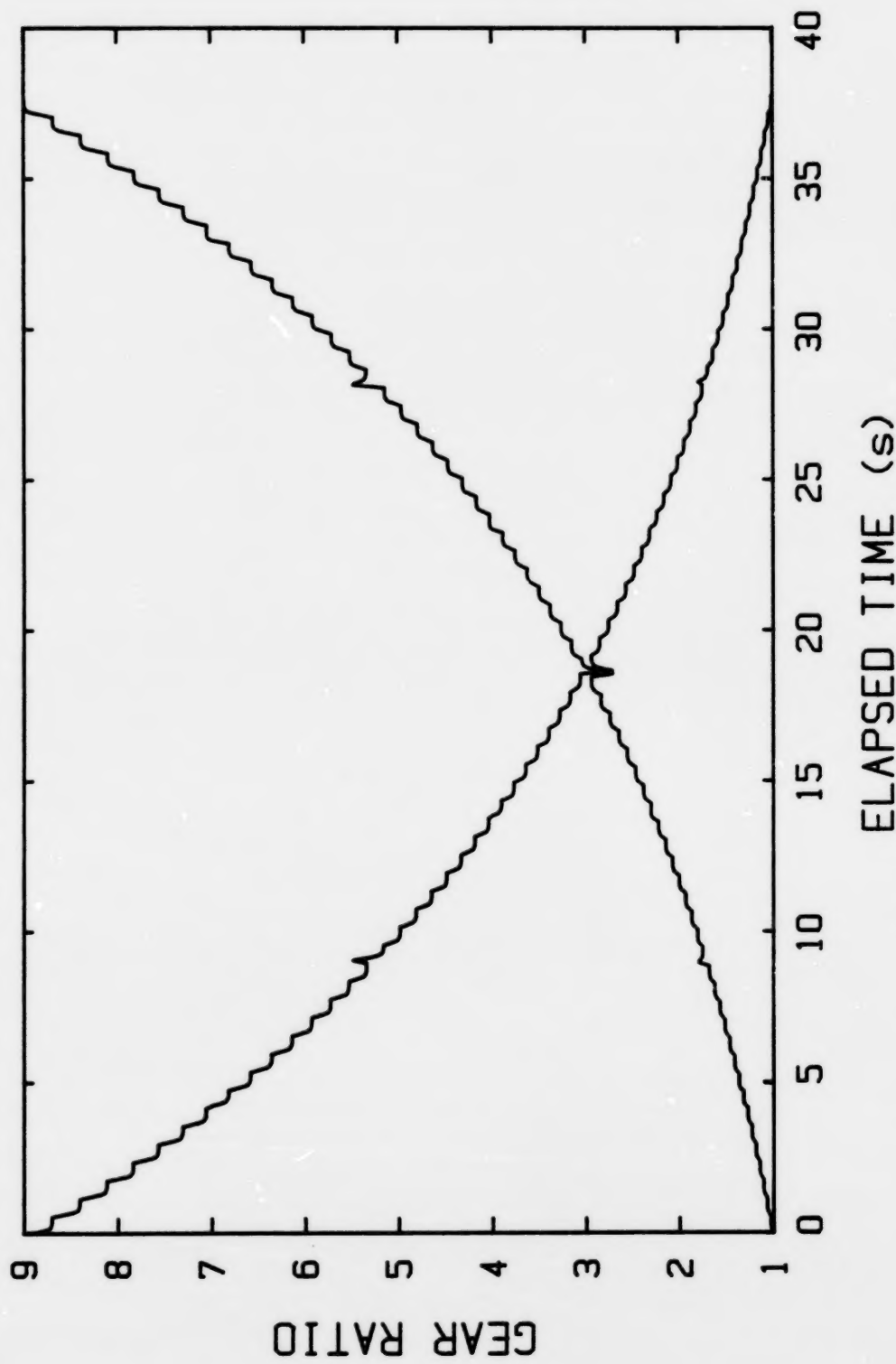


Fig. 3 Ideal Upshift and Downshift Patterns.

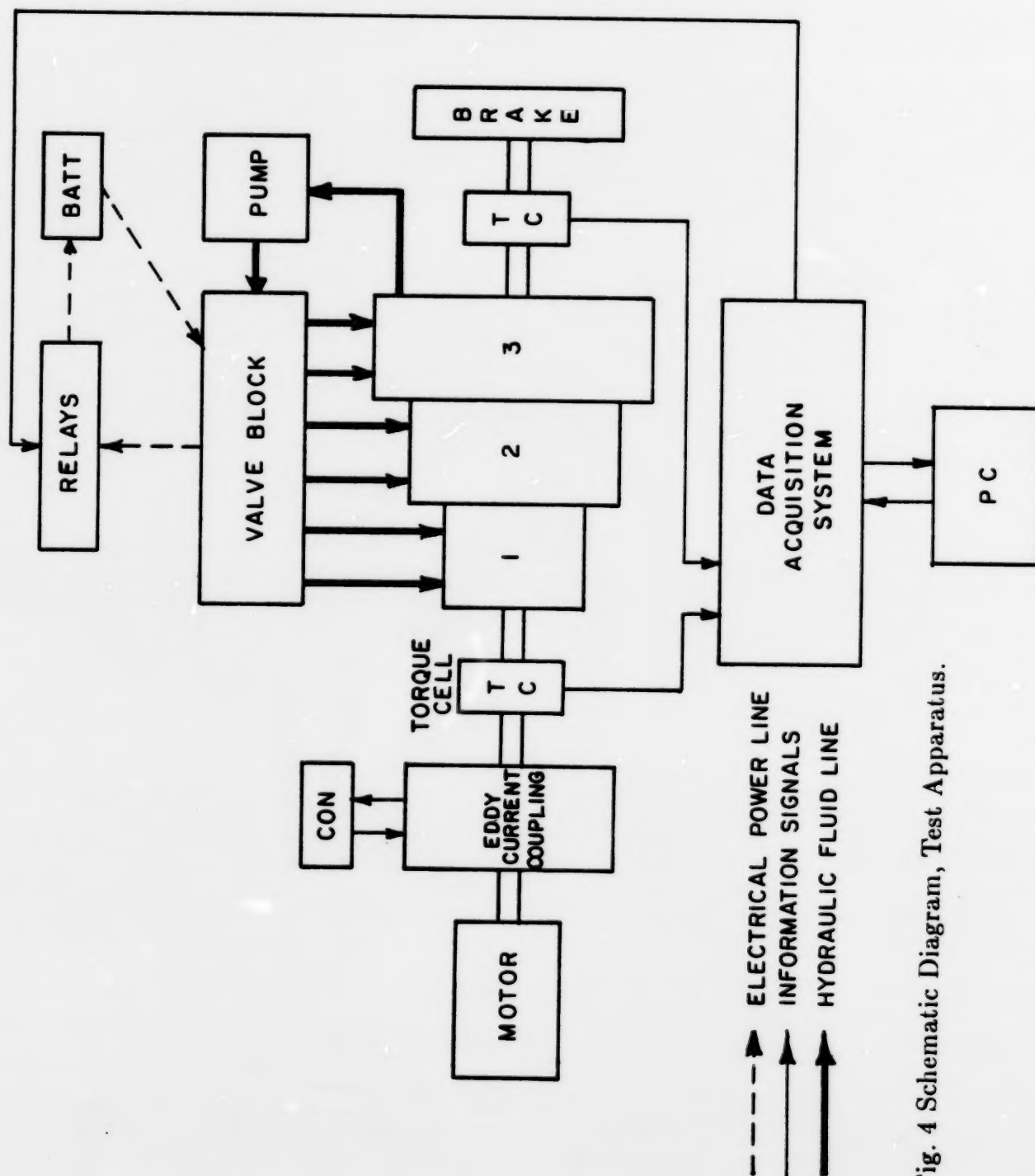


Fig. 4 Schematic Diagram, Test Apparatus.

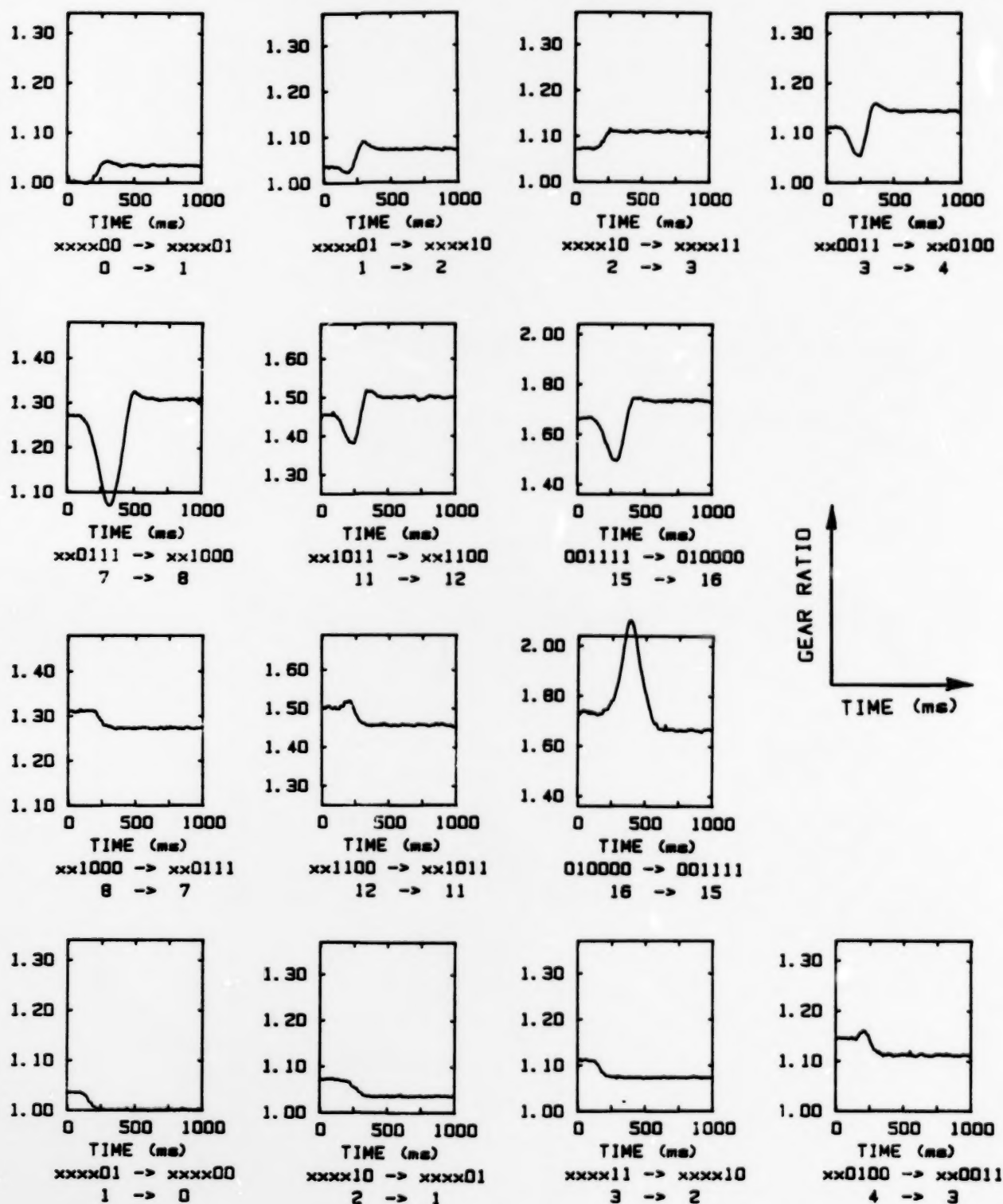


Fig. 5 Untimed Shifts for Classes 1 to 7.

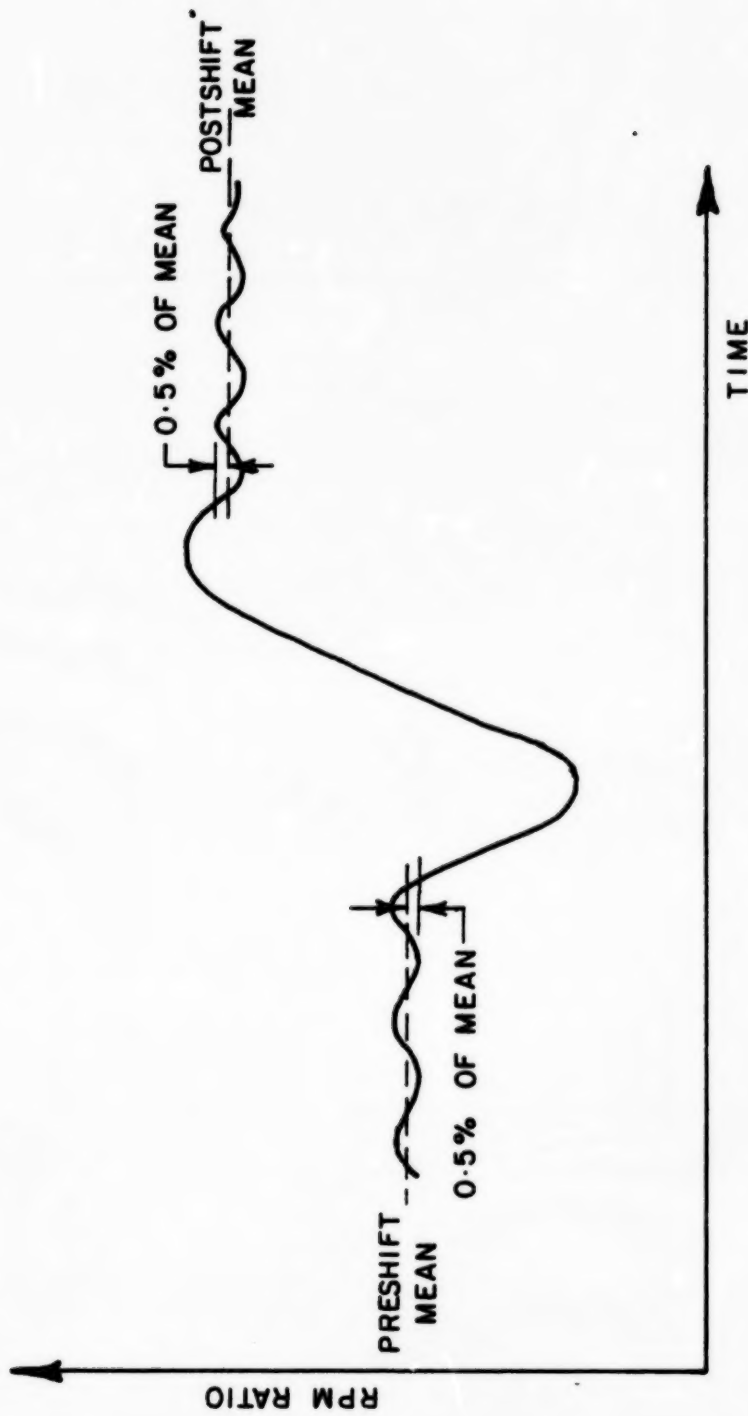
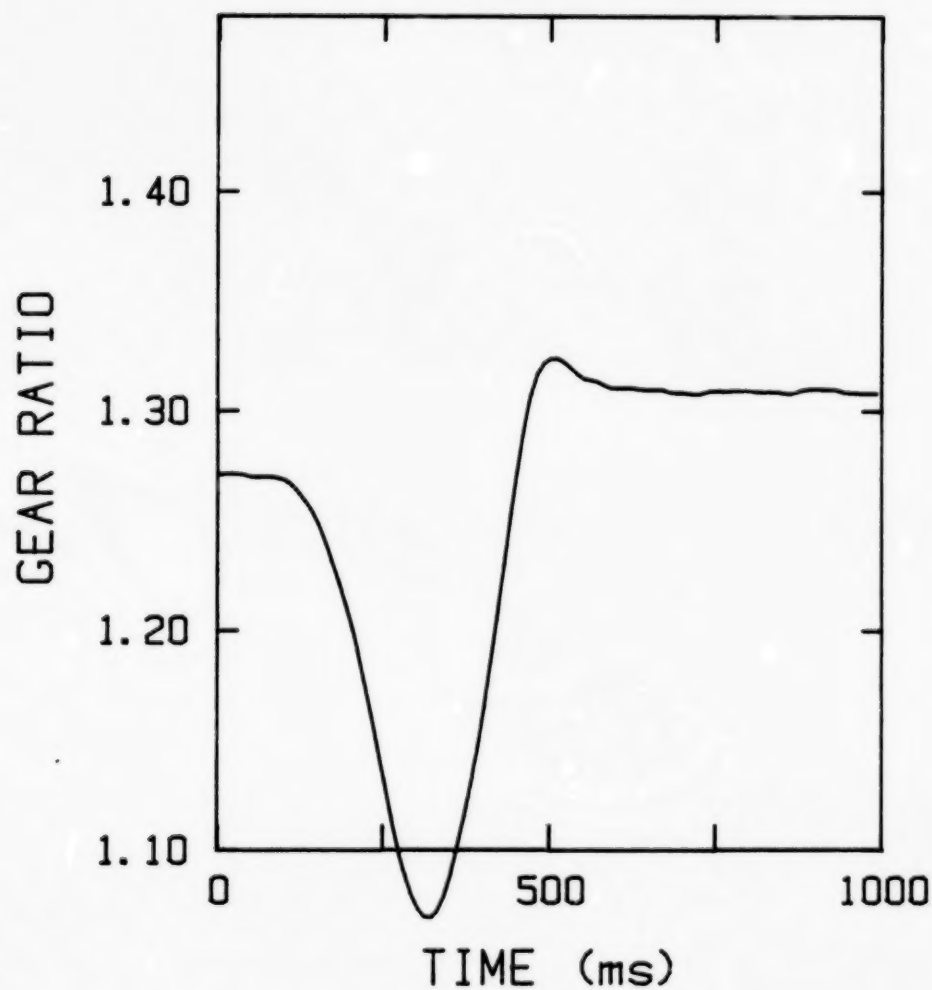


Fig. 6 Shift Completion Criterion.

UNTIMED SHIFT

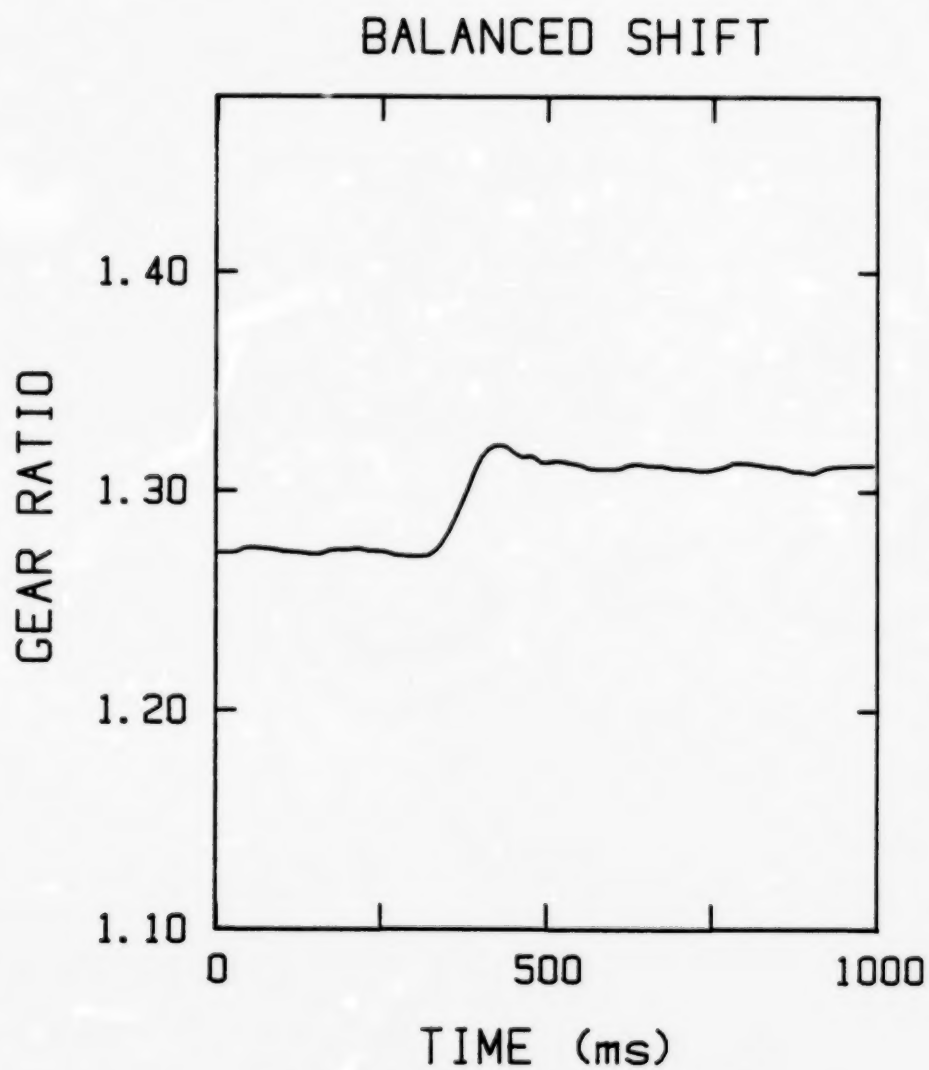


xx0111 -> xx1000

7 -> 8

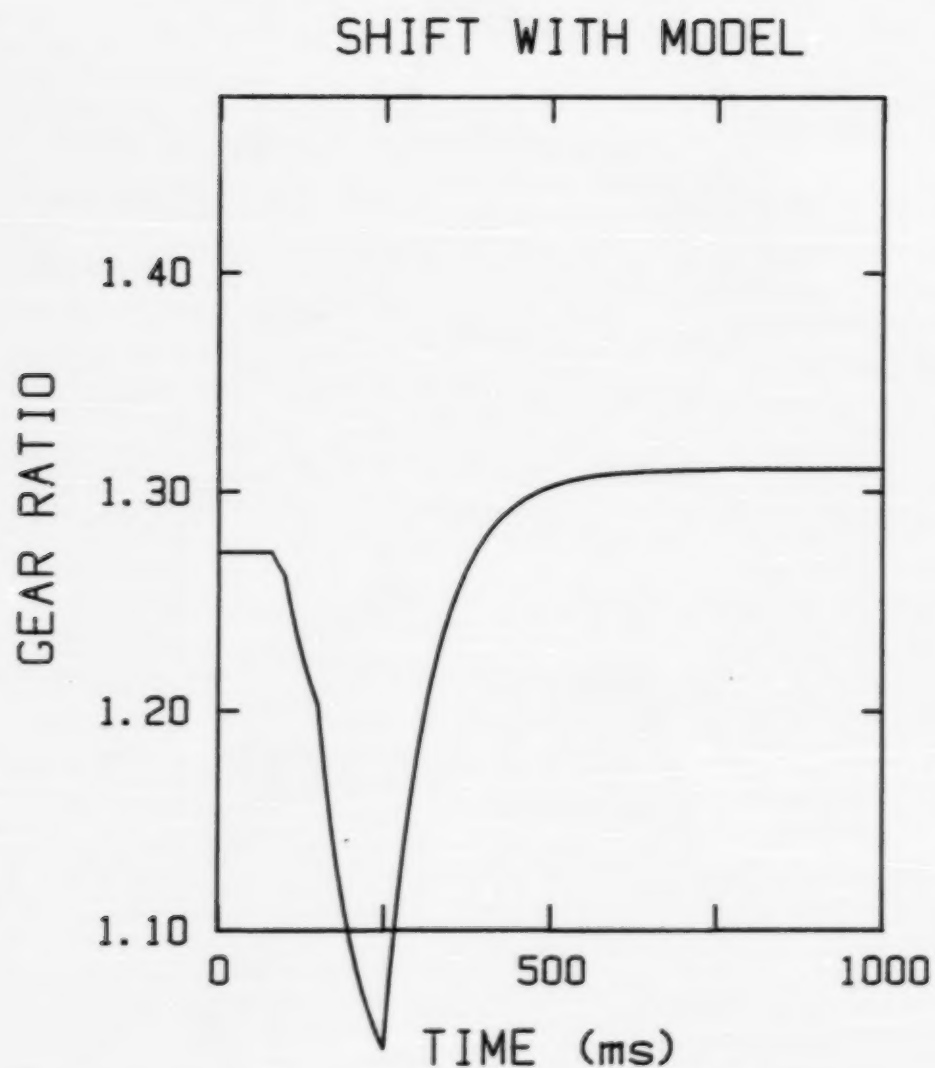
1.272 -> 1.31

Fig. 7 Untimed 7-8 Shift.



xx0111 -> xx1000
7 -> 8
1.272 -> 1.31

Fig. 8 Balanced Clutch Action, 7-8 Shift.



xx0111 -> xx1000

7 -> 8

1.272 -> 1.31

Fig. 9 Kinematic Model, 7-8 Shift.